

D-Zero Run2B Silicon Stave Cooling Model and Test Results

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1. Abstract

A cooling study of a prototype mechanical stave for the D-Zero Run2b upgrade was done at the Silicon Detector Facility to determine the effectiveness of the coolant passage design and to test the accuracy of the stave thermal model.

2. Summary

The test validates the proposed coolant operating parameters and design of the stave's coolant passage. The goal of the design is to maintain a maximum silicon temperature of 0°C. Silicon temperatures in Layers 2 to 5 will be readily managed with the intended core design and operating parameters of the coolant. Additionally, the testing confirms the accuracy of heat transfer calculations and finite element analysis used to design the cooling passage. The models predict appropriately conservative silicon temperatures. Silicon temperatures at other operating conditions can be confidently predicted by analysis without the need for further bench testing.

3. Stave Core Cooling Passage

To keep stave mass to a minimum, the core of the stave is fabricated from foam and lightweight plastics. A drawing of the D0 stave is shown in Figure 1. Figure 2, an exploded view of the stave, reveals the construction of the stave's core. The cooling tube is U-shaped and includes inlet and outlet nozzles at one end for supply of coolant. The cooling passage is formed from extruded PEEK (polyether-etherketone) tubing. PEEK has a high tolerance for radiation and is readily formed into the rectangular cross-section tube geometry designed for the core. The cooling tube has inside dimensions

of 1.70 mm x 6.81 mm and a wall thickness of 0.10 mm. The flow path is about 1.2 m long. The tube is sandwiched between 0.08 mm thick Kapton MT skins. A thin adhesive layer exists between the tube and the skins and between the skins and the bottom side of the silicon.

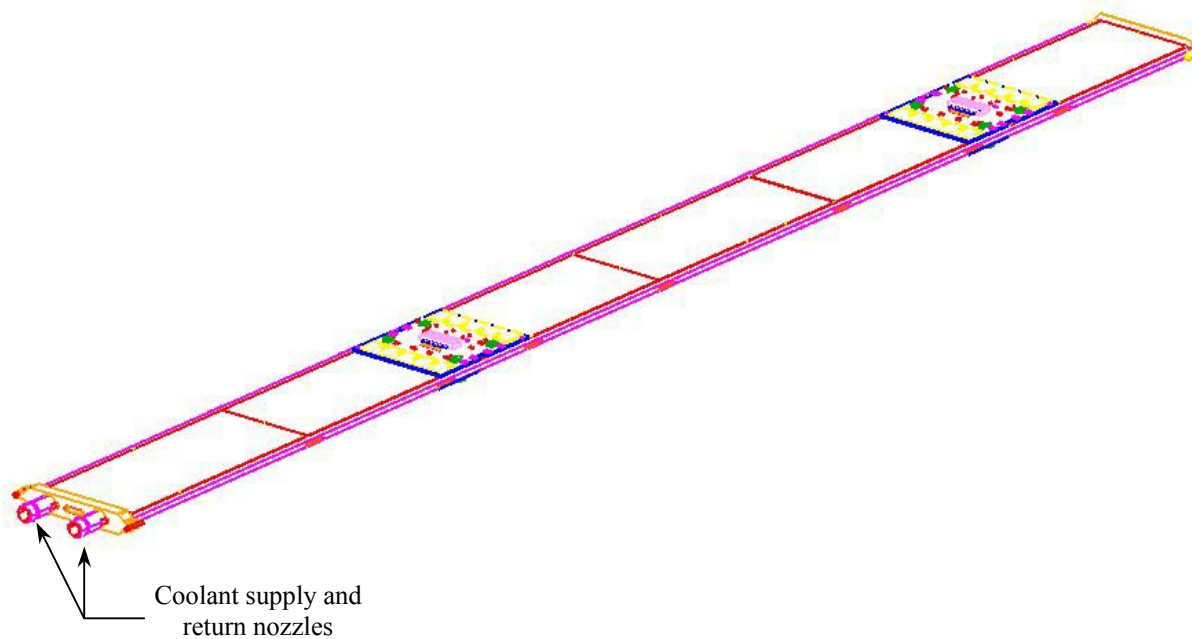


Figure 1- D-Zero Run2b Stave (C-channels not shown)

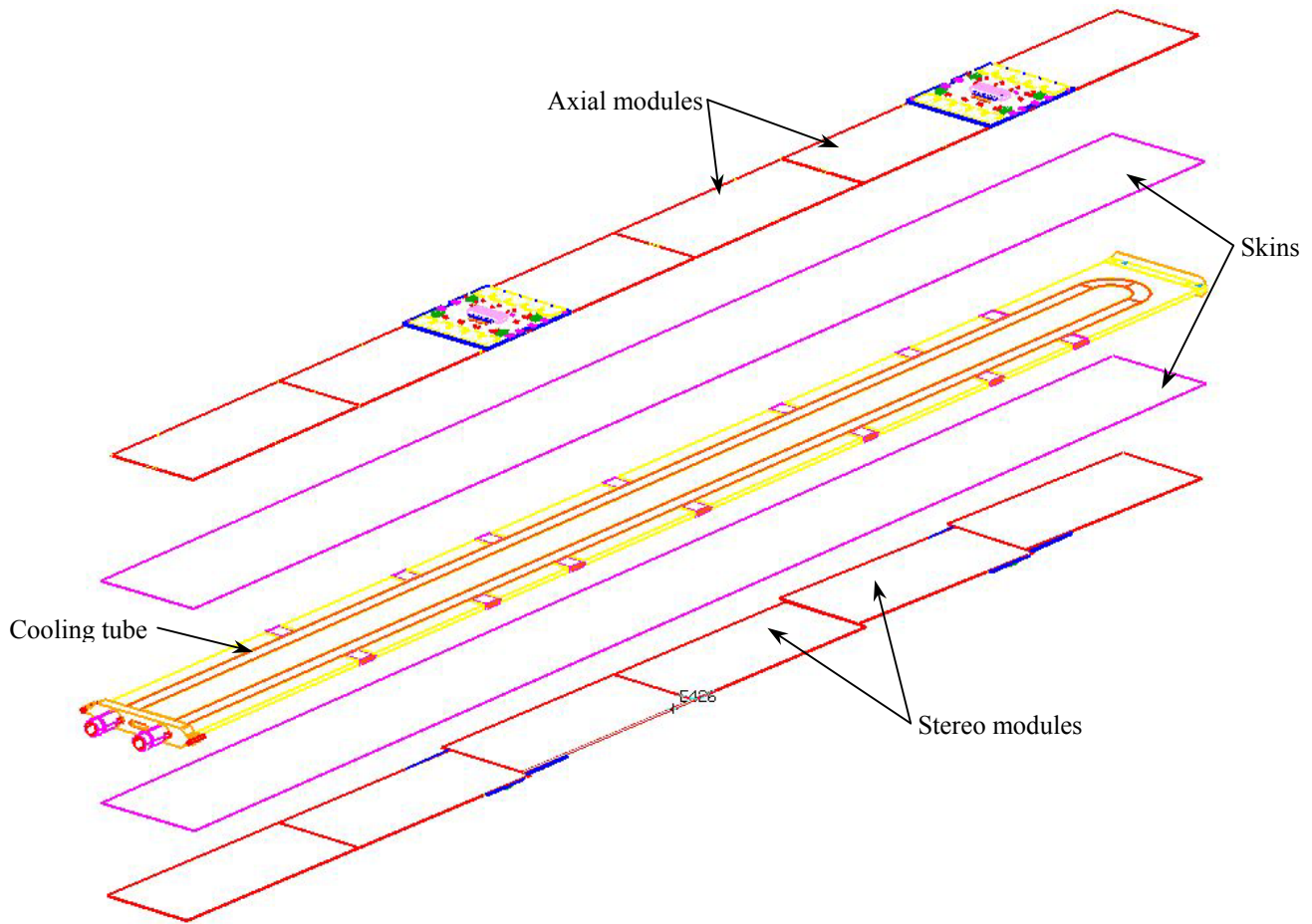


Figure 2- Exploded view of D0 stave

4. Coolant Parameters

A mixture of ethylene glycol and water will serve as the coolant for the Run2b detector. The expected operating temperature of the coolant is -15°C and the mixture is 41% EG by volume. The freezing point of this mixture is -25.8°C , providing margin for protecting against freeze-out of water from the solution in the chiller's evaporator and the possibility of operating at colder temperatures.

5. Fluid Flow

The existing piping and chiller system currently serving the Run2a silicon detector at D-Zero will be used for Run2b. A careful study of the existing piping system shows that at the expected coolant operating parameters, the system will provide adequate cooling capacity and permit a pressure drop of 3 PSI across the staves. The existing pumps will provide enough flow capacity to permit every stave to

be plumbed in parallel to achieve maximum cooling of the staves in every layer. Yet the opportunity exists to daisy-chain inner layers with outer layers to develop a radial temperature gradient within the detector, operating inner layers colder for higher radiation doses and outer layers slightly warmer to minimize the cooling effect on the adjacent fiber tracker.

Given a 3 PSI differential across a single stave, the coolant flow rate is calculated using correlations for rectangular tubes found in Kakac et al.¹ The expected flow rate is 0.196 LPM. The flow in the tube is laminar. The velocity profile develops quickly, with the hydraulic entrance length calculated to be less than 20 mm.

6. *Heat Transfer Film Coefficient Analysis*

In laminar flow, as is present within the stave cooling tube, the convection coefficient is not proportional to the fluid velocity. It is a function of the shape of the cooling passage, the thermal conductivity of the fluid, and certain boundary conditions that are explained in greater detail here. Kakac includes correlations for determining the convection coefficient within rectangular ducts. Correlations are provided for hydraulically developed and developing flow, thermally developed and developing flow, uniform heat flux at the cooling tube surface or uniform surface temperature. For the stave, as stated in the previous paragraph, the hydraulic flow profile develops quickly so a fully developed velocity profile is assumed. Neither a condition of uniform heat flux or uniform surface temperature exists along the cooling tube. Underneath the hybrids, there may be more of a uniform heat flux condition while further away from the hybrids, the silicon approaches uniform temperature. For the analysis, all parameters (Nusselt number, convection coefficient, thermal entrance length) are calculated as the mean of the two values determined for both thermal boundary conditions (uniform heat flux and uniform surface temperature).

With the temperature profile fully developed (steady-state thermal boundary condition), the convection coefficient calculates to be:

$$h = 670 \text{ W/m}^2\text{-K}$$

Thermally developed, heat transfer around full perimeter of the tube

¹ Kakac, Shah, and Aung, Handbook of Single-phase Convective Heat Transfer, John Wiley and Sons, 1987.

As noted above, the result assumes that heat is transferring into the fluid through all four walls of the rectangular tube. The literature presents further refinement for calculation of the convection coefficient for cases where any surface(s) around the duct perimeter are adiabatic (perfectly insulated). The stove cooling tube picks up heat from the silicon only through the top and bottom surfaces of the tube. The sidewalls of the cooling tube are in contact with the foam in the core so they do not contribute to the available cooling surface area. Therefore they are considered adiabatic. The resulting thermal boundary condition around the inside perimeter of the cooling tube has a positive effect on the heat transfer. The convection coefficient increases. For this tube geometry with adiabatic sidewalls, it calculates to:

$$h = 805 \text{ W/m}^2\text{-K}$$

Thermally developed, heat transfer through top and bottom surfaces of tube

Again, the assumption above is that the temperature profile is fully developed. This convection coefficient is 20% higher than the coefficient calculated assuming heat transfer through all four sides of the cooling tube. At this tube geometry, the adiabatic sidewalls comprise 20% of the surface area of the tube. This ‘loss’ in heat transfer surface area is mathematically ‘recovered’ by a 20% gain in the film coefficient.

Before the temperature profile fully develops, better heat transfer (higher heat flux) exists as a result of the steeper temperature gradient from the tube wall to the tube center. As the temperature profile develops along the flow path, the magnitude of the local film coefficient decreases until the steady-state, fully developed condition is reached. The convection coefficients presented above are for this steady-state condition. Kakac presents correlations for calculating the local convection coefficient in a rectangular duct as a function of distance from the duct entrance. The entrance length is a measure of how far a slug of fluid passes through a tube before its temperature profile fully develops. For this tube geometry the entrance length is 0.94 m. The full path length of the stove cooling tube is 1.2 m. So the coolant will not achieve a fully developed temperature profile until it is just about to exit the stove. Table 1 shows the local convection coefficients calculated along the tube length. Local film coefficients are calculated underneath each hybrid. The average value of the local convection coefficient is also shown.

Table 1- Local heat transfer coefficient along tube length

Hybrid Location		Nusselt number			Local film coefficient, h [W/m^2-K]
In detector CS	Along flow path				
z [mm]	x [mm]	Nu,x_T	Nu,x_qs"	Nu,x avg	
400	200	5.95	7.58	6.77	930
100	500	5.03	6.16	5.60	769
100	700	4.86	5.86	5.36	737
400	1000*	4.44	5.33	4.89	670

*Fully developed

Average-> 777

The correlations for developing flow provided by Kakac do not include corrections for the adiabatic wall conditions. Table 2 below summarizes the mean film coefficients calculated for the stave.

Table 2- Mean heat transfer coefficient [W/m²-K]

Tube wall boundary condition	Fluid Temperature Profile	
	Fully Developed	Developing
Four tube walls conducting	670	777
Side tube walls adiabatic	805	???

In Table 2, the upper left film coefficient is most conservative. The upper right and lower left numbers were calculated from assumptions that more closely represent actual conditions within the stave. The two numbers are essentially the same. The lower right quadrant would be the best approximation of the stave's operating conditions but correlations for that case were not located. From the previous paragraphs, it seems reasonable to assume that the film coefficient for a duct with adiabatic sidewalls and a developing temperature profile is at least as large as the conditions represented by the cell either to its left or above it.

7. *Finite Element Analysis*

The finite element model of the stave is represented in Figure 3. Given the geometrical and thermal symmetries the model has been consequently simplified to one quarter of the structure. The cooling tube inner size is 6.8mm × 1.7mm (.268" × .067"). The turn around is about 43mm from the z=0 end.

The PEEK tube wall is assumed to be 0.1 mm thick and has been modeled only for the area in contact with the glue layer under the sensor. The poor thermal conductivity of the Rohacell justifies the approximation. The coolant has been assumed to be at a bulk temperature of -14°C. The model was solved for two film coefficients: 790 and 695 W/m²K. The larger number is the average of the two cases summarized in Table 2 (upper right and lower left quadrants) that best approximate the boundary conditions in the stave. The second number was run as both a sensitivity test of the model and to place a conservative upper limit on silicon temperatures. It is arbitrarily 12% smaller.

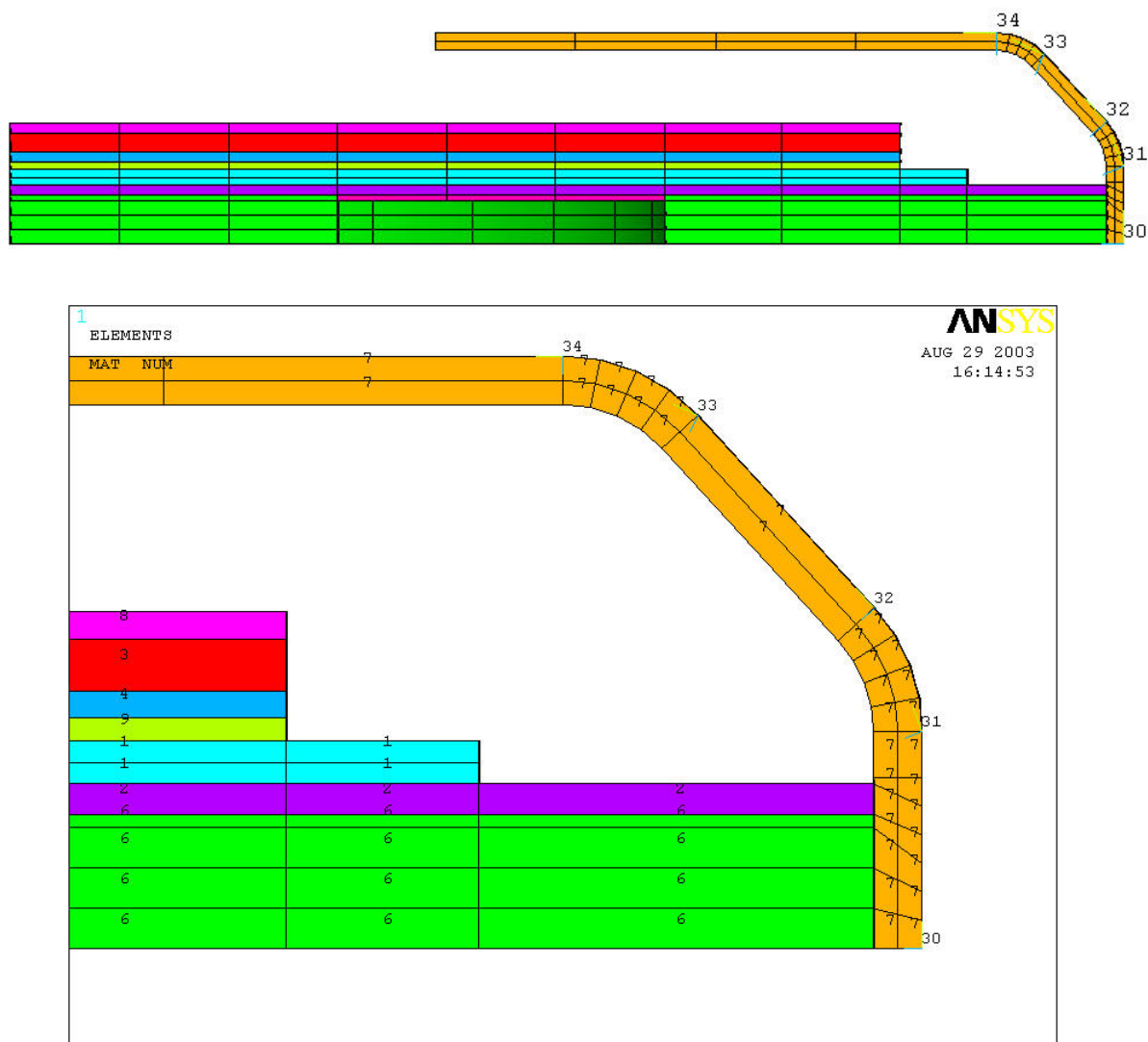


Figure 3 – Front and detailed view of the stave FEA model

Table 3 shows thermal conductivity and thickness of the materials used in the FEA model.

Table 3 - Material thermal conductivities and thicknesses used in FEA

Material	number in Figure 3	color in Figure 3	Thermal conductivity [W/m K]	Thickness [μm]
silicon	1	Light blue	163.0	320
Kapton + Epoxy layer	2	violet	0.22	225
Beryllia	3	red	248.0	380
Glass	4	blue	1.38	200
PEEK	-	fuchsia	0.25	100
Rohacell ²	6	green	2.9×10^{-3}	1000
Carbon Fiber K139	7	orange	$K_x = 79.8$ $K_y = 17.9$ $K_z = 0.24$	360
Gold + glass layer	8	magenta	1.38^3	210
Patterned glass + epoxy layer	9	Light green	0.47	170

8. *Test Results*

A prototype mechanical stave was tested to verify the cooling system design. The stave is assembled with blank silicon modules, outfitted with blank BeO hybrids. Photographs of the test setup are shown in the figures below.

Electrical heaters were attached to the blank hybrids to simulate the heat load of the SVX chips. Two of the heaters can be seen in Figure 5. The voltage and current supply to the heaters were both measured. The stave was instrumented with nine RTD's and one RTD was installed in both the cooling line supply and return connections on the stave. Supply and differential pressures of the coolant were also measured at the stave inlet.

² The total Rohacell thickness is 2mm. Only 1mm has been modeled because of the symmetry.

³ Assumed equal to the pure glass.

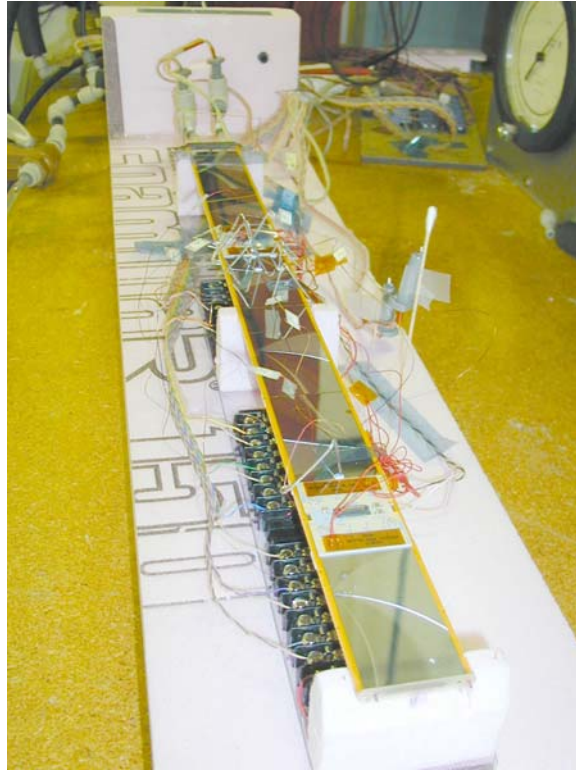


Figure 4- Photograph of thermal test

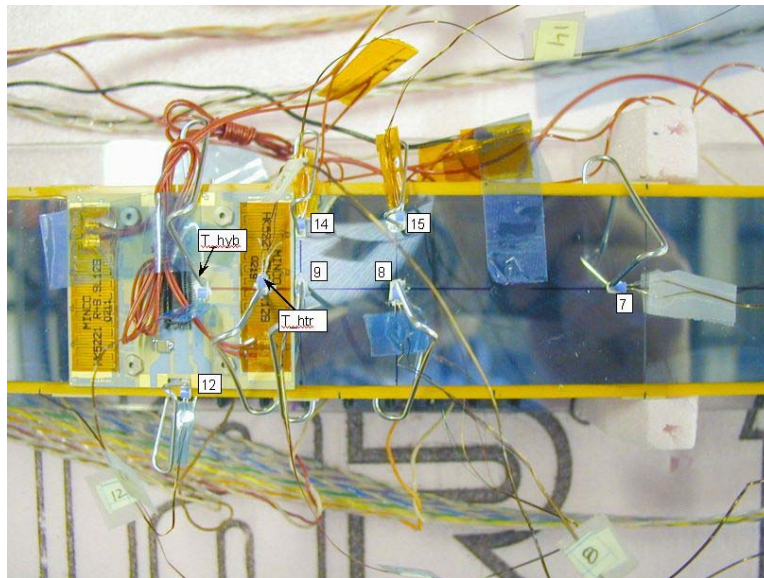


Figure 5- RTD locations. Looking at 20/20 hybrid, stereo side. RTD 6 is adjacent to 10/10 hybrid (not pictured)

An insulating box was built to control ambient heat loads. Thermal tests were performed with the box sealed. Measurements with an open box give a worst-case upper limit on the silicon's operating temperatures due to the additive ambient heat load. The closed-box measurements are likely a better simulation of the detector's environment when installed at D-Zero. Tests were done during the cold winter months when ambient humidity was low. No condensation collected on the stave during the open box measurements.

The ethylene glycol mixture was verified by measurement with a Misco digital refractometer. The mixture measured 42.5% EG by volume. The flow rate through the stave was measured with a stopwatch and a 500 mL graduated cylinder. The recorded flow rates were used in energy balance calculations. The measurements are consistent with expected flow rates but an accurate comparison cannot be made because of the test setup. Previous studies showed that flow rates predicted by the published correlations are accurate. The differential pressure across the stave was held constant at approximately 3.25 PSID for the tests (3 PSI target for the stave and allowing another 0.25 PSI for the jumper tubes and fittings for temperature measurement connected between the stave and the pressure gage).

The plot in Figure 6 below shows the temperature data collected. Regions of the test are labeled to show when the heaters were on and off. The numbers inserted into the temperature plot correspond to the numbers shown in Figure 5 above. The plot shows that data was collected with the coolant inlet temperature at -10°C and -15°C . The expected coolant operating temperature is -15°C . 'T_{fluid}' in the chart is the average of the inlet and outlet temperature. 'T_{inf}' is the air temperature recorded inside the box.

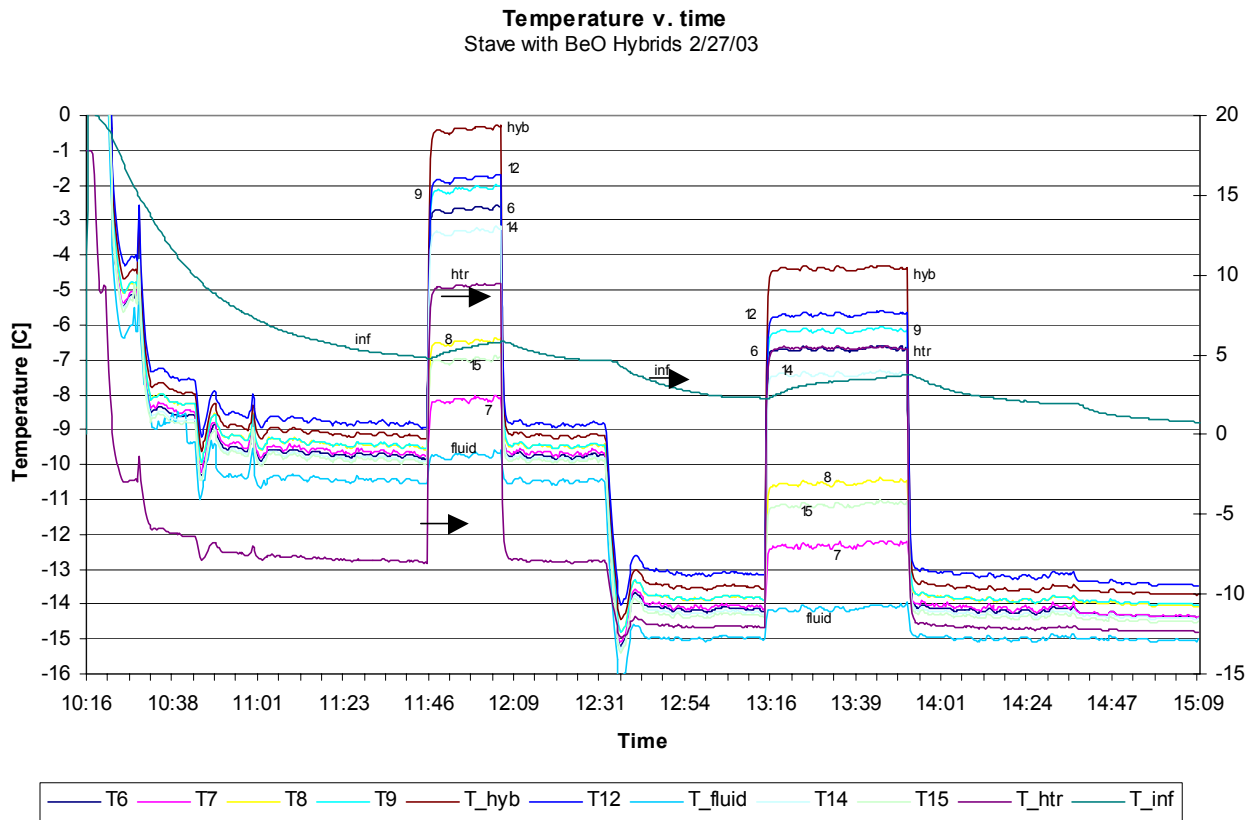


Figure 6 - Temperature data

Figure 7 is a plot with some of the data manipulated. Figure 8 is an overlay of output from the finite element analysis and test data. The analysis predicts that the silicon temperatures will be kept below 0°C , as the design parameters require. The tests results validate the model and illustrate that the assumptions made in its construction are adequately conservative.

Silicon to Fluid Differential Temperature
BeO hybrid on dummy Silicon - 2/27/03

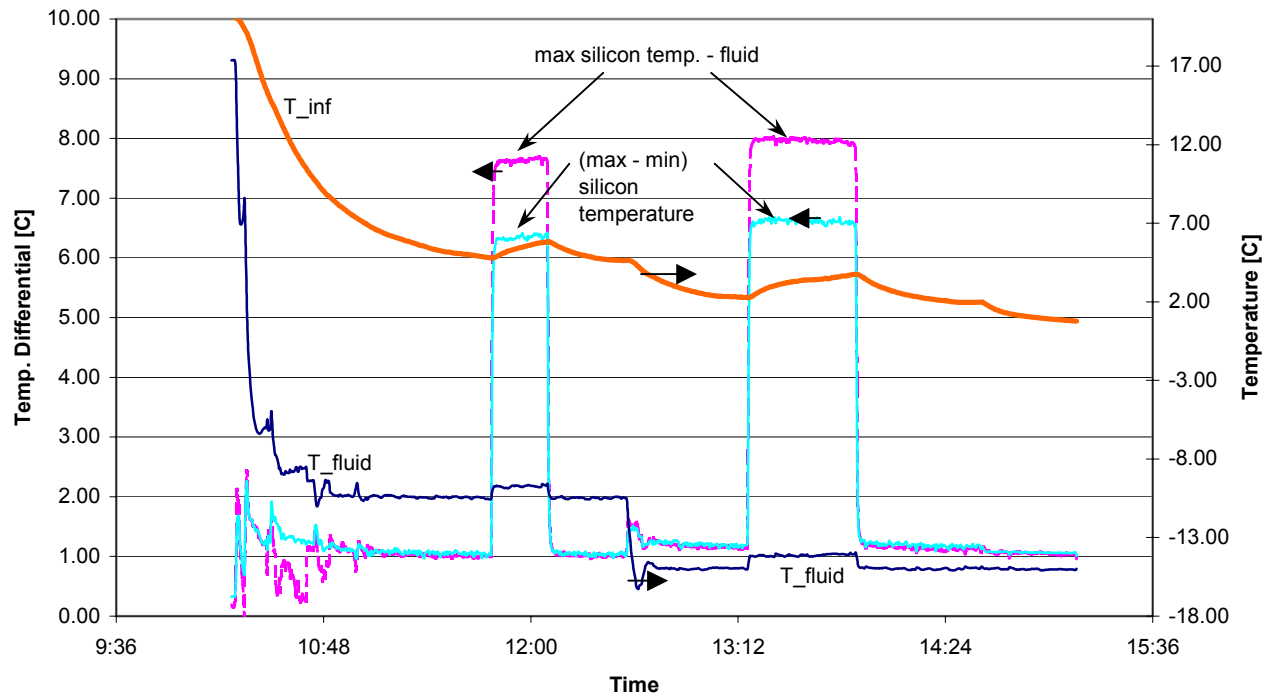


Figure 7 - Temperature differentials

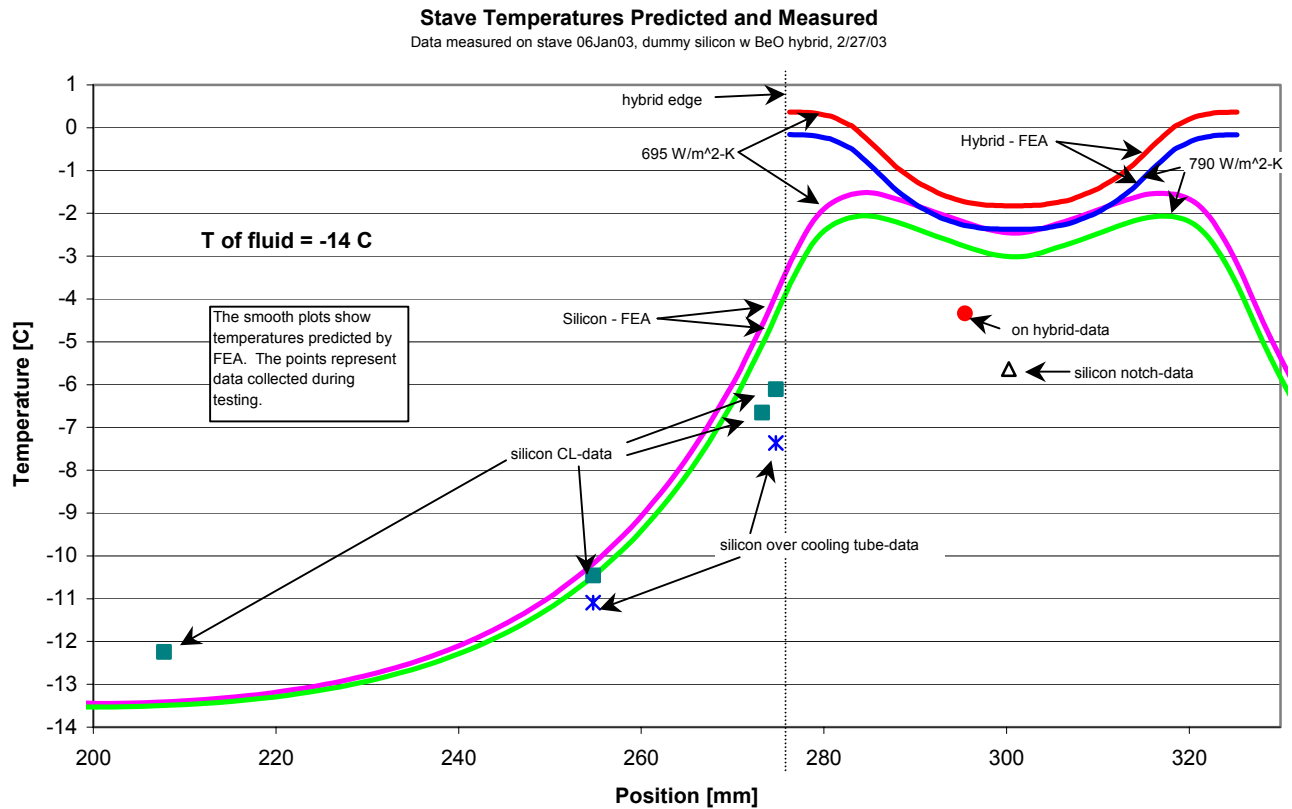


Figure 8- Comparison of FEA and test data

9. Summary

The design of the cooling tube and the operating parameters of the coolant will manage silicon temperatures in Layers 2-5 as required. Silicon temperatures will not exceed 0C. The finite element model is validated. If other operating parameters need be explored, FEA will adequately predict temperatures so that further bench testing is not required.